

Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behavior

By R. T. KNAPP,¹ PASADENA, CALIF.

This paper describes the technique of determining the complete operating characteristics of a hydraulic machine such as a centrifugal pump or a turbine, together with a method of presenting these characteristics in a convenient manner on a single diagram. The characteristics of a modern, high-head, high-efficiency pump are analyzed and presented in the manner proposed. The use of these complete characteristics for the prediction of the behavior of the machine during operating transients is discussed and the analytical background is presented. The assumptions involved are investigated and experimental checks of their validity are offered. The interrelationships between the hydraulic characteristics of the machine and the pipe line are indicated.

IF THE possible operating conditions of hydraulic-turbine and centrifugal-pump installations are compared, it soon becomes apparent that the pumps are subject to much wider and more involved variations than are the turbines, especially during the transient states of starting, stopping, or emergency operation. In turbines the direction of flow and the direction of rotation are always the same, even in case of a breakdown of the machine itself or trouble in the penstock and auxiliary equipment. Thus the machine performance always lies in the quadrant of normal turbine operation, and since its hydraulic characteristics are very well-known in this quadrant, it is a comparatively straightforward matter to predict the complete performance during any possible transient condition. On the other hand, under similar conditions with a pump installation, the flow can completely reverse direction, as can the rotation. The machine in this case ceases to be a pump, and after passing through a zone of energy

dissipation, becomes a runaway turbine. This great variation in performance gives rise to many questions, such as the runaway speed of the machine as a turbine, the time of reversal, the magnitude of the accelerating forces, the effect on the surge cycle in the discharge line, the maximum and terminal reverse rates of flow, and so on. Unfortunately these questions are very difficult to answer, because, although the hydraulic performance of the machine is well-known as long as it is acting as a pump, comparatively little study has ever been made of the performance as an energy dissipator or as a turbine.

The objective of this study has been, therefore, to explore these little-known regions of performance and to attempt to use the resulting information to answer some of these important questions.

PREVIOUS INVESTIGATIONS

In 1931 Kittredge and Thoma (1)² published an article on "Centrifugal Pumps Operated under Abnormal Conditions." This paper described experiments carried on with a small pump for the purpose of obtaining performance characteristics from which the behavior of the pump during sudden changes of operating conditions could be predicted. In these experiments the pump was operated under conditions of negative head, delivery, and speed, in addition to the normal range of performance. As an outgrowth of the work a series of investigations was undertaken in the hydraulics laboratories of the California Institute of Technology, under the direction of the author.

In the fall of 1931, Boothe and Lewis (2) started a preliminary investigation on a $1\frac{1}{2} \times 10$ -in. single suction pump. Although the results were very interesting, it was felt that the pump was too small and the efficiency too low for them to be completely typical of modern installations.

In the spring of 1932, two 4-in. high-head high-efficiency pumps were made available through the generosity of the Byron-Jackson Company. These pumps were installed in the summer of 1932 and work was carried on with them for about two years. The first results were outlined by Haynes and Sauermann (3) in 1933, and in 1934 a more complete presentation was made by the present author (4). The study also furnishes most of the background for this article.

DETERMINATION OF COMPLETE CHARACTERISTICS

(A) *Laboratory Equipment.* The objectives of the program demanded that the pump under test be so installed that it could be operated under all possible conditions of flow, head, and speed, both as a pump and as a turbine. Therefore, of the two available, the one with the lowest head and capacity was selected as the test pump, while the other was designated as the service or supply pump and was connected so that it could deliver either to the suction or the discharge lines of the test pump. The test pump itself was connected to a Sprague electric dynamometer which was capable of being operated either as a motor or a generator at any speed up to 3500 rpm and in either direction of

² Numbers in parentheses refer to the Bibliography at the end of the paper.

¹ Associate Professor of Hydraulic Engineering, California Institute of Technology. Jun. A.S.M.E. Professor Knapp was graduated with a B.S. degree from the Massachusetts Institute of Technology in 1920, and received his Ph.D. degree in mechanical engineering from the California Institute of Technology in 1929. He was designer for the Gay Engineering Corporation 1920-1921, and instructor at the California Institute of Technology from 1922 to 1930. Since 1923 he has been in charge of the power-plant laboratory at the California Institute of Technology, and has been in charge of the hydraulic laboratory since 1927. He acted as consulting engineer for the Riverside Cement Company, Los Angeles, Calif., from 1927 to 1929. He was awarded the A.S.M.E. Freeman Scholarship in Hydraulics in 1929. At present he is also consultant for the Metropolitan Water District of Southern California and the Los Angeles County Flood Control District.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

rotation. It had a capacity of 100 hp or about twice the maximum required by any operating condition of the pump. Fig. 1 is a diagram of the complete equipment. It will be noted that a spray pond and auxiliary pump are provided for cooling the system. This was necessary because the combined power input of the test and service pumps went as high as 150 hp, and since the system was a closed one with a comparatively small volumetric capacity, such a power input would have caused a rapid rise in temperatures if arrangements had not been made to dissipate it.

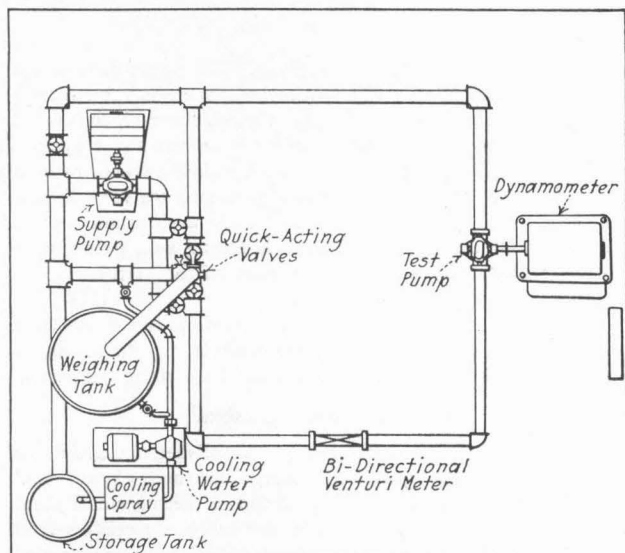


FIG. 1 PIPE LAYOUT FOR PUMP TESTS

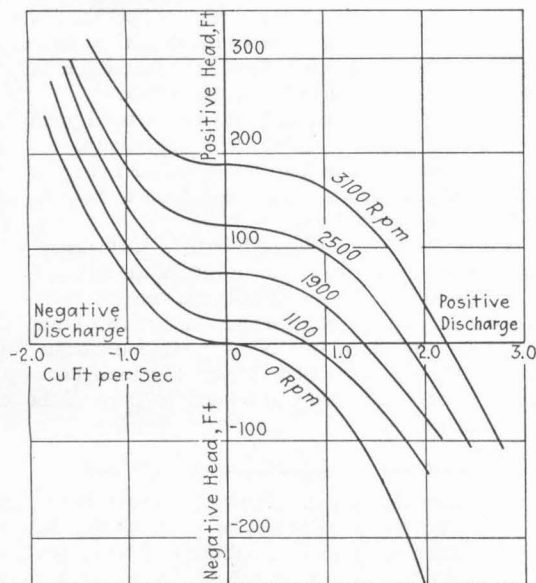


FIG. 2 POSITIVE-ROTATION HEAD-DISCHARGE CURVES

(B) *Instruments.* The basic method of measuring the rate of flow was by use of the weighing tank and synchronized chronograph. The working secondary standard was a bidirectional venturi tube, calibrated in place. This tube was constructed in the laboratory, and consisted of two symmetrical long-taper cones with piezometer rings in the center and at each end. Speed was measured with a magnetic-clutch revolution counter

or by a revolving contact which recorded each tenth revolution on the chronograph drum. Low pressures were measured by means of a six-foot mercury manometer, while special ground-piston, fluid-pressure scales reading to $1/10$ lb per sq in. were constructed for the high-pressure measurement. Torque was measured directly by the scales on the cradle dynamometer. At all points care was taken to insure the accuracy of the operations.

(C) *Experiments.* The schedule of tests consisted of a series of constant-speed runs, both in the normal and in the reverse direction. For each speed the discharge was varied from a maximum-negative to a maximum-positive value. By negative discharge it is meant that the flow was from the discharge to the inlet of the pump. The range of discharges was about the same for all speeds, varying from about -200 to +150 per cent of the normal for the pump when operating at 3100 rpm. It should be

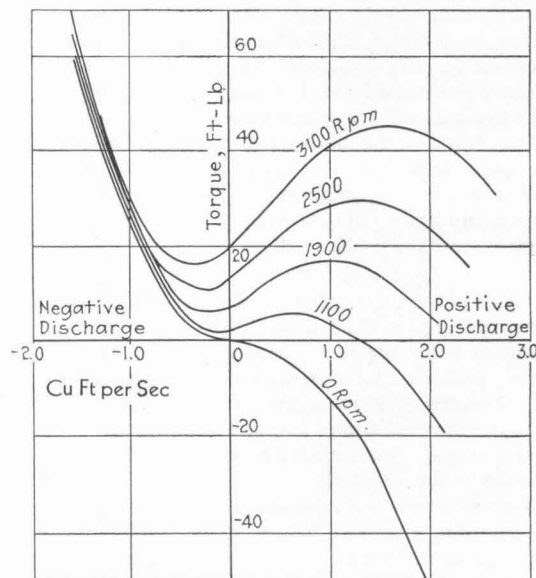


FIG. 3 POSITIVE-ROTATION TORQUE-DISCHARGE CURVES

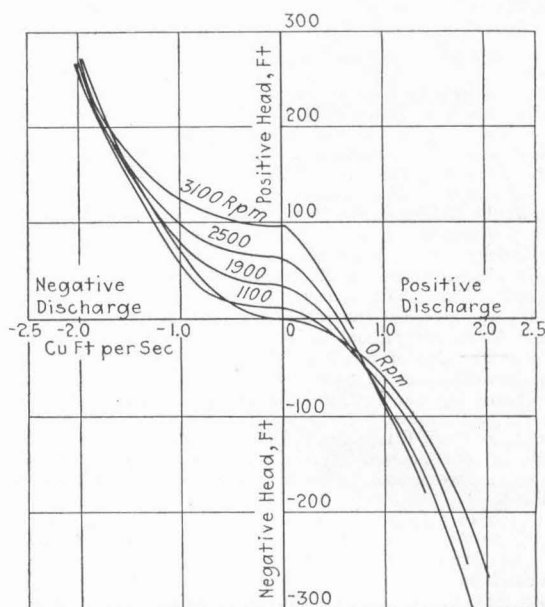


FIG. 4 NEGATIVE ROTATION HEAD-DISCHARGE CURVES

noted that one of the constant-speed runs in the series was that of zero speed. For this run the impeller and shaft were locked to a torque arm and torque readings taken directly as the flow was varied over the usual range.

(D) *Constant-Speed Curves.* The results of these runs were first plotted in the series of constant-speed curves shown in Figs. 2 to 5 inclusive. Figs. 2 and 4 are the head-discharge curves for positive and negative rotation, while Figs. 3 and 5 show the torque-discharge relations. If these curves are examined, several interesting points will be noted; for example, (a) there are no discontinuities even at points where head, discharge, or torque reverses sign, (b) the zero-speed curve is smooth and has the characteristic shape of the family.

(E) *Complete Characteristics Diagram.* In the papers of Kirtledge and Thoma (1), previously referred to, the final results were presented in two series of curve sheets. The first set showed the variations in head and power plotted against the discharge, while the speed was held at either a constant positive or negative value. The second set plotted the same variables against speed while the discharge was held at given constant values. The first set was of course analogous to Figs. 2 to 5 of this article.

When the final presentation of the results of the current test was considered, it was suggested by Prof. Th. von Kármán of

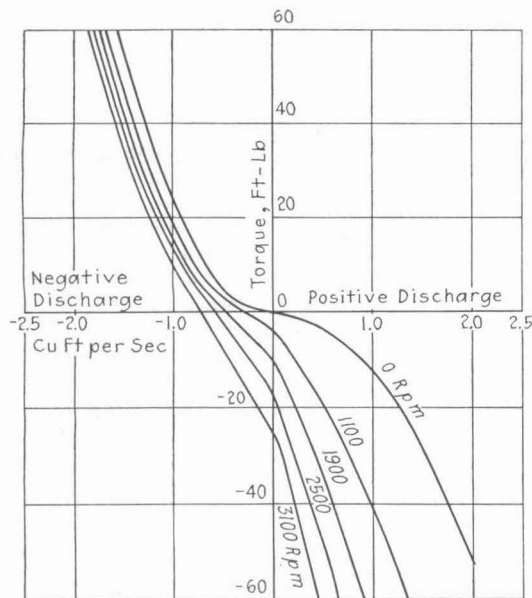


FIG. 5 NEGATIVE-ROTATION TORQUE-DISCHARGE CURVES

the Institute that a more comprehensive and useful picture of the complete performance of the machine would be obtained if all of the results were presented on a single four-quadrant diagram having as coordinates the discharge and the speed. The merits of this particular diagram are more easily seen if a brief study of its characteristics is made before it is used. Fig. 6 is an explanatory chart for such a diagram, and shows the way the various zones of operation separate themselves. It will be seen that there are two zones of pump operation, the normal one in the first quadrant and the abnormal or reverse rotation one in the second quadrant. Likewise there are two zones of turbine operation, the normal one in the third quadrant, and the abnormal or outward-flow one in the first quadrant. Each of these zones of possible useful operation is separated from the others by zones of energy dissipation in which no useful work is done either on or by the fluid. For example, the entire fourth quadrant is such a

zone, and separates the zone of normal pump operation from that of normal turbine operation. Since all possible conditions of operation may be represented on this chart it may be named the "complete characteristics diagram" of the hydraulic machine tested. It is of course equally applicable either to a pump or a turbine.

Fig. 7 is the complete characteristics diagram of the 4-in. double-suction pump tested. The data are presented as series of

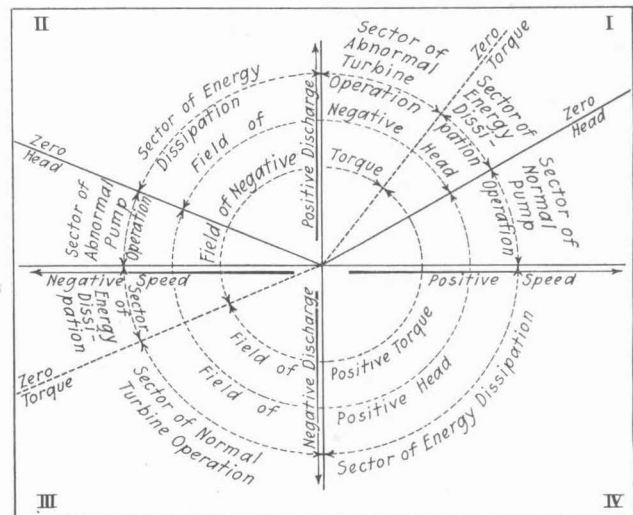


FIG. 6 EXPLANATORY CHART FOR COMPLETE CHARACTERISTICS DIAGRAM

contour curves of constant values of head and torque, the full lines being lines of constant head and the dotted ones lines of constant torque. It is interesting to observe that both pairs of zero-head and zero-torque lines are practically straight. They, of course, form the asymptotes for the corresponding families of contour curves. Each family shows reasonable agreement with the similarity laws, i.e., the entire family can be approximately calculated from any single curve. There are some deviations, however, in the zones of abnormal operation and energy dissipation. These deviations are probably caused principally by incipient or developed cavitation in some part of the machine. Therefore, these zones of the diagram are somewhat sensitive to the value chosen for the base pressure for the particular condition of flow being tested. From this point of view it would appear advisable to standardize on the normal pressure existing at the pump inlet under the condition of the proposed installation, if the diagram is to be used for calculating transient behavior.

For some purposes it is desirable to add a third set of contours to this diagram, i.e., lines of constant efficiency. They would exist only in the four zones of useful operation, since the efficiency is zero throughout all four zones of energy dissipation. In the abnormal pump and turbine zones all efficiencies are very low, but high efficiencies of the same order of magnitude were found in both the normal pump and turbine zones.

Many of the critical operation points can be read directly from the diagram. For example, for one given head the speed at which pumping ceases and flow reverses is given by the intersection of that constant-head line with the speed axis, which is the line of zero flow; the negative flow at which the machine reverses its direction is determined by the intersection of the same constant-head line with the discharge axis, which is the line of zero speed; and the corresponding turbine runaway speed is located by the crossing of the zero-torque line by this same constant-head con-

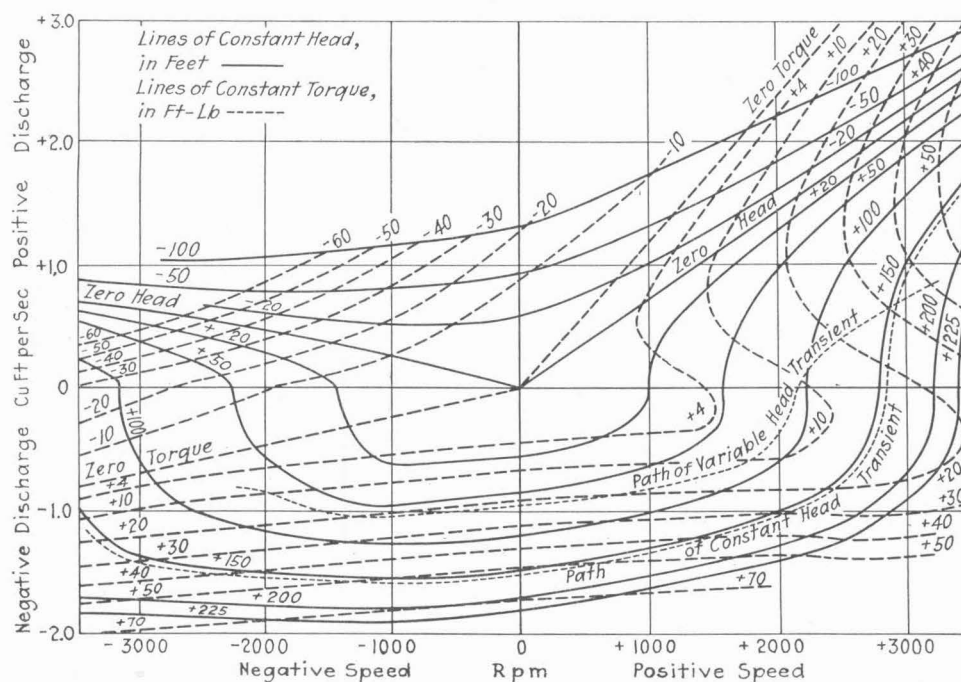


FIG. 7 COMPLETE CHARACTERISTICS DIAGRAM FOR 4-IN. DOUBLE-SUCTION PUMP

tour. The maximum reverse flow and the corresponding speed are determined by the point of tangency of a horizontal line with this head contour.

PREDICTION OF TRANSIENT BEHAVIOR

(A) *Basic Principle.* As has been previously stated, the complete characteristics diagram, Fig. 7, covers all possible conditions of operation of the machine within the ranges of speed and discharge shown. Therefore, if there is a change in the operating conditions it must be possible to plot the path of this change on the diagram. Consider a simple example: Suppose the test pump operated at 3200 rpm and discharged water into a reservoir 150 ft above the suction supply, through a submerged outlet pipe line so large that both friction losses and possible surges could be considered negligible. Then, no matter what the pump did the head would remain constant. Thus the operating point would always remain on the 150-ft head line on the diagram. If the driving power were removed, the operating point would move down the line out of the first quadrant, through the fourth and into the third quadrant until it intersected the line of zero torque at a negative speed of approximately 3600 rpm (turbine runaway) which would be the new point of equilibrium. Simultaneous values of discharge, speed, and torque can be read from the diagram for every point along the constant-head line. Since the driving torque was removed, the values of torque indicated by the diagram must be supplied by the deceleration of the rotating mass of the system. If the polar moment of inertia of this rotating system is known, it is possible to calculate the time-speed relations existing during the change. If any other speed-head path had been followed, similar calculations could have been made.

The usefulness of this method is that it offers a way of calculating the time characteristics of such events as normal starting and stopping of a pumping plant, abnormal shutdown due to power failure, the behavior of the pump in case of shaft breakage, or other emergency conditions arising in plant operation. Of course, it alone does not give the entire picture, for in the normal installa-

tion the surge pressures in the discharge line during transients are of major importance. However, the complete characteristics diagram of the machine furnishes the information needed to calculate the surge, and the surge pressures are necessary in order to plot the speed-head path on the diagram. This apparently indicates that a step-by-step calculation would be necessary for the complete determination of the performance. The results should be well worth the effort, for, with the information they would furnish, the designer would be in position, for example, to determine with surety whether or not relief or quick-closing valves would be necessary to the operation of the system, and if so their complete time characteristics.

By the use of the graphical method for water-hammer problems described by Schnyder (6), Bergeron (7), and Angus (8), this step-by-step solution can be greatly simplified when studying surge conditions in pump-discharge lines.

(B) *Assumptions Involved.* The use just proposed for the complete characteristics diagram involves major assumptions that must be thoroughly understood and borne in mind during the application. The principal assumption upon which most of the others depend, is that the instantaneous performance of the machine for any given set of momentary conditions occurring during a transient is identical with the steady-state performance for those same operating conditions. In this assumption two others are implied, (a) that two or more types of flow cannot exist within the machine for one given set of operating conditions, even momentarily, and (b) that the momentary accelerating forces exerted on the fluid *within the machine* during the transient are small in comparison to the forces required for normal steady operation at that particular state. If these assumptions do not hold true within the limits of accuracy desired for the calculations or if corrections cannot readily be applied to bring them within these limits then the method is not applicable. It was, therefore, decided to try to carry out an experimental test of the method and the assumption for some simple cases of transient behavior that could be studied within the limitations of the laboratory equipment. However, before these experiments are

described, the analytical background of the prediction of transient behavior will be discussed in more detail.

(C) *Analytical Background.* The equation of motion of a rotating system is,

$$T = I \frac{d\omega}{dt} \dots \dots \dots [1]$$

where

- T = unbalanced torque applied to system
- I = moment of inertia of the system about the axis of rotation
- ω = angular velocity
- t = time

if ω be replaced by its equivalent $\frac{2\pi N}{60}$ this becomes

$$T = \frac{\pi}{30} I \frac{dN}{dt} \dots \dots \dots [2]$$

where N = revolutions per minute.

This may be integrated to obtain

$$t_2 - t_1 = \frac{\pi}{30} I \int_{N_1}^{N_2} \frac{1}{T} dN \dots \dots \dots [3]$$

If the torque is known as a mathematical function of speed it may be possible to integrate this analytically, but if the relation is only known empirically it is always possible to evaluate the integral graphically. The latter is the more normal case. The

procedure is simply to plot $\frac{\pi I}{30T}$ as a function of N . The area under the curve between any two values of speed, N_1 and N_2 , is equal to the time required for the speed to change from N_1 to N_2 . If the entire integral curve is plotted it will give the speed as a function of the time.

The method just described is precisely that used to obtain a prediction of the time-speed relations during a pump transient from the known speed-head relations and the complete characteristics diagram. Of course, as mentioned under section (A) of this division of the article, the speed-head relationship is often an implicit function of the discharge-line characteristics and the machine characteristics. In such a case it would probably be necessary to plot and integrate the differential curve step by step. After the integral curve has been plotted, giving the speed-time relationship, one more piece of information can be obtained. The speed values of this curve are identical with those on the complete characteristics diagram, following the line of the known speed-head relation. But along this line, each speed corresponds to a definite value of the discharge. Consequently values from the two diagrams can be combined to give the discharge-time curve for the transient change.

(D) *Experimental Verification of Transient Predictions.* A study of the method of transient analysis just presented suggested the most direct type of experimental check. This was to carry the pump through a known change of operating conditions, experimentally determining the actual speed-time relationship, and comparing this with the prediction calculated according to the method described from the observed speed-head relationship. If the two agreed, this would be direct evidence of the validity of the major assumption of the method.

The simplest way of carrying on these experiments appeared to be to start with the test and service pumps both operating and discharging into a common line. If the power supply of the test pump was cut off then the supply pump would furnish the pres-

sure necessary to maintain the head. The test pump would decelerate, reverse, and come up to speed as a runaway turbine.

The regular instrumentation permitted such experiments. The primary readings necessary were the speeds and times. These were automatically recorded on the chronograph record of time vs. total revolutions. The instantaneous speed was determined accurately by measuring the number of revolutions recorded in a small interval of time. Two types of runs were made. By controlling the service-pump discharge valve the head on the test pump was kept constant independent of the test-pump speed. By the other method the setting of the discharge valve was not changed, and, as the test pump slowed down, the discharge pres-

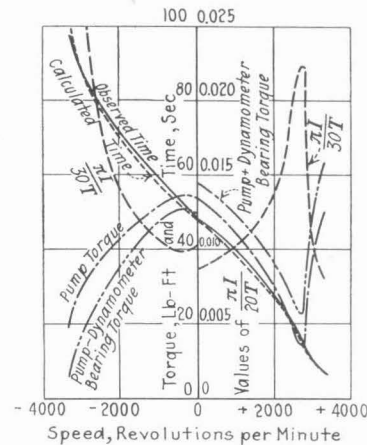


FIG. 8 TRANSIENT SPEED-TIME RELATIONS
(Constant-head run.)

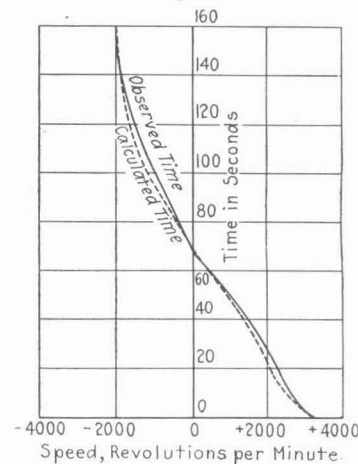


FIG. 9 TRANSIENT SPEED-TIME RELATIONS
(Variable-head run.)

sure fell until the equilibrium condition was reached where the test pump ran as a runaway turbine on the water supplied by the service pump. For this case the speed-head data were simultaneously recorded by observers.

The method of predicting the transient relationship was the same for both tests. First the head-speed path was plotted on the complete characteristics diagram directly from the readings taken during the experiment. From this path the torque-speed relationship was plotted. During the actual run an additional torque acted on the rotating system. This was the dynamometer bearing friction, the value of which had previously been carefully determined. This friction always opposed the direction of mo-

tion. The action of the water opposed the motion until the runner had been brought to a standstill, then produced an acceleration until the runaway speed was reached. The resultant torque on the system was thus the sum of the two from the initial point to the point of zero speed, and the difference of them from that point to the point of equilibrium runaway condition.

Fig. 8 shows a constant-head run. The speed-head path may be seen in light dotted lines in Fig. 7. The curve of Fig. 8 labeled pump torque shows the speed-torque readings from Fig. 7. The curve labeled pump and dynamometer torque shows the addition of the dynamometer bearing friction. The reciprocal of this curve, multiplied by the constant $\frac{\pi}{30} I$, was then plotted. The integral curve of this is the dotted speed-time curve. The solid speed-time curve shows the actual experimental values.

Fig. 9 shows the predicted and observed speed-time curves for a case where the head varied during the transient. The path of this run may also be seen in light dotted lines in Fig. 7. These curves are typical of the group of check runs that were made. In all cases the agreement between the predicted relationship and the observed values was better than could reasonably be expected.

In one respect, these check runs do not completely satisfy all questions. Due to the large moment of inertia of the dynamometer armature, all the transient runs which it was possible to make have a relatively slow rate. In the faster one shown, Fig. 8, it is seen that it took about 48 sec until the point of re-

versal of rotation, and about 100 sec until the equilibrium runaway speed was approached. This is considerably longer than would be the case with the average field installation. On the other hand, the agreement here is so good that a considerable additional error would be tolerable before the usefulness of the results would become impaired.

CONTINUATION OF INVESTIGATION

During the spring of 1934, while the last of these tests were being completed, a new hydraulic-machinery laboratory was being designed and constructed at the California Institute of Technology, in cooperation with the Metropolitan Water District of Southern California. It was put into operation in August of that year and since that time an intensive program of study of the problems of high-head pumping plants of the Colorado River Aqueduct has been carried on. A description of this laboratory by the author (5) may be found in the Transactions. A part of this work has been the determination of the complete characteristics of several of the model pumps tested in the laboratory. Fig. 10 shows one example of these diagrams which is published through the courtesy of the Byron-Jackson Company and the Metropolitan Water District. The pump, the characteristics of which are shown, is a much larger one than that used in the foregoing work, requiring about 300 hp when operating at the point of best efficiency against a head of 300 ft. It represents the best current pump practice, since in the model it showed a test efficiency of over 92 per cent. It is anticipated that in the near future

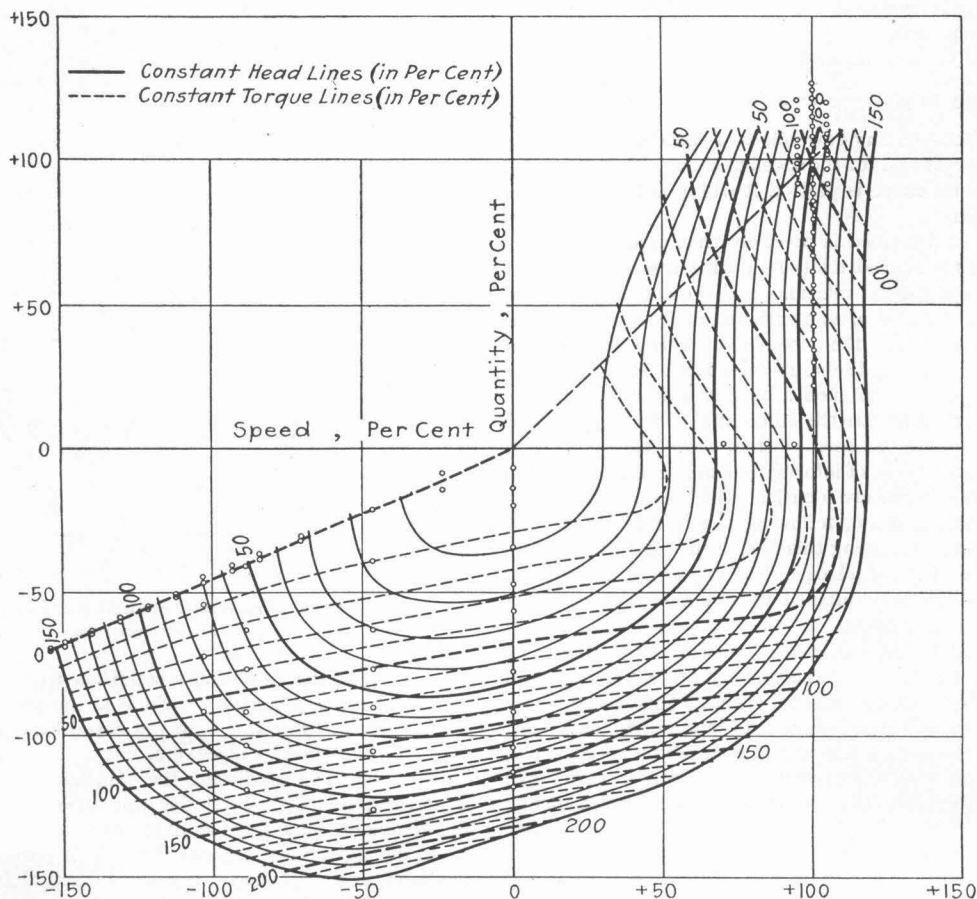


FIG. 10 COMPLETE PUMP CHARACTERISTICS
(Single-suction intake model; specific speed, 1700 rpm.)

more complete tests will be made of this and other large-scale models of varying specific speeds, and also that other transient check runs at higher accelerations can be carried out to test more thoroughly the validity of this method of calculating transients.

ACKNOWLEDGMENT

In addition to the men whose names are mentioned in references (2) and (3) and to the Byron-Jackson Company and the Metropolitan Water District, the author wishes especially to acknowledge the assistance of Dr. Geo. F. Wislicenus, who worked diligently with him in installing the apparatus and instruments in the original laboratory, who helped direct the experimental work, and who collaborated with him on the analysis and application of the results.

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